

Experimental investigation of a micro-combined cooling, heating and power system driven by a gas engine

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Received 2 February 2005; received in revised form 12 April 2005; accepted 28 April 2005

Available online 18 July 2005

Abstract

An experimental investigation of the performance of a micro-combined cooling, heating and power (CCHP) system is described. The natural gas and LPG-fired micro-CCHP system uses a small-scale generator set driven by a gas engine and a new small-scale adsorption chiller, which has a rated electricity power of 12 kW, a rated cooling of 9 kW and a rated heating capacity of 28 kW. Silica gel–water is used as working pair in the adsorption cooling system. The refrigeration COP of the adsorption chiller is over 0.3 for 13 °C evaporation temperature. The test facility designed and built is described, which supplies better test-rig platform for cooling, heating and power cogeneration. Experimental methodology of this system is presented and the results are discussed. An energetic analysis of micro-CCHP system is performed as well. The overall thermal and electrical efficiency is over 70%.

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Keywords: Cogeneration; Experiment; Generation; Gas engine; Chiller; Adsorption system; Water; Silica gel; Performance; COP

Etude expérimentale sur le refroidissement, le chauffage et la generation d'énergie entraînée par un moteur à gaz

Mots clés : Cogénération ; Expérimentation ; Génération ; Moteur à gaz ; Refroidisseur de liquide ; Système à adsorption ; Eau ; Gel de silice ; Performance ; COP

1. Introduction

Combined cooling, heating and power (CCHP) for buildings is a concept that encompasses the local production of electricity for applications along with the utilization of

waste heat produced during the power production process. The waste heat can be made available for heating and/or cooling applications using thermally activated technologies such as absorption chillers and adsorption chillers. The advantage of CCHP for buildings is the high primary energy efficiency of the system, the environmental benefits and economic feasibility [1–7]. This is accomplished as a result of the utilization of the heat made available from electricity generation as well as elimination of losses due to transmission and distribution because the power is produced on site.

Cogeneration is a well-known technology for energy conservation in industry and in commercial buildings.

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Nomenclature

CCHP	combined cooling, heating and power
LPG	liquefied petroleum gas
COP	coefficient of performance
PER	primary energy rate
CHP	combined heat and power
CCP	combined cooling and power
Q	heat transfer rate (kW)
P	electrical power (kW)
T	temperature (°C)
HE	heat exchanger
$C_{p,w}$	specific heat of water ($\text{kJ kg}^{-1} \text{°C}^{-1}$)
m	mass flow rate (kg s^{-1})
r	electricity to thermal heat recovered ratio of the system
η	efficiency
θ	heat load supplied to useful thermal heat recovered ratio of the micro-CCHP system

Superscripts and subscripts

HP	heat and power supplied separately
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CP	cooling and power supplied separately
CHPS	cooling, heat and power supplied separately
in	inlet
outlet	outlet
jacket_water	jacket cooling water of the engine
exhausted_gases	exhausted gases of the engine
el	electricity
th	thermal
total	thermal and electrical
heat_source	heat source water of adsorption chiller
hot	total recycled hot water of the micro-CCHP system
chilled	chilled water of adsorption chiller
ad_chiller	adsorption chiller
conv	conventional independent system
loss	heat loss
el_chiller	electrical air conditioner
boiler	boiler
heat_load	heat load supplied by the micro-CCHP system

However, limited research has been conducted for energy conservation in small scale buildings, such as residential users. Internal combustion engines and combustion turbines are widely recognized as important power generation technologies for CCHP applications today, especially for large-scale applications such as district energy systems (10 MW utility size applications) and industrial cogenerations (5–100 MW applications) [8–12]. Small scale gas engine have many advantages, such as its availability in small sizes, fast start-up and shutdown capability, good part load operation and high electrical efficiency, ability and flexibility to run with different working mediums (LPG, natural gas). The waste heat recovery will not affect the mechanical energy output from the engine [13–15].

Normally absorption chiller is considered for utilization in cogeneration, due to its availability of products driven by steam or hot water, or even direct fired with natural gases or oil. For the steam or directed fired absorption chillers, the cooling COP is usually higher than 1.2 as double effect absorption cooling is used [16]. But the market available absorption chillers are usually at rated cooling power over 100 or 200 kW, which makes micro CCHP systems not ready, the research work on CCHPs are usually for large systems, not laboratory experimental work, but field tests. The idea of micro cogeneration in small-scale buildings has been thought reasonable with the development of small adsorption chillers (≤ 10 kW) driven with low-grade thermal energy (60–90 °C). For example the 90 °C hot water is available from the exhausted gases and jacket cooling in gas engines or diesel engines which is reasonable to produce summer cooling [17–19], and the recovered waste heat from engine jacket cooling is nearly the same as

that from exhausted gases. If the both recovered heat can be applied, the waste heat utilization will be very efficient. Recently, a prototype machine of adsorption chiller with capacity 10 kW is developed by Institute of Refrigeration and Cryogenics of Shanghai, Jiao Tong University (SJTU) [20,21]. The tests have shown its cooling COP reaches 0.3–0.4 with a heat source of 60–95 °C.

In this paper, a micro-CCHP with a small gas engine and adsorption chiller is presented, and the description of the test facility designed and built to evaluate the performance of micro-CCHP modules is described. Experiment methodology of this system is presented and the results are discussed. An energetic analysis of micro-CCHP system is performed as well. This kind of micro-CCHP system test facility supplies better test-rig platform for cooling, heating and power cogeneration. Sufficient research on energy management and operational optimization of CCHP systems could be made by using it. The energy management method of the system could be used in any large-scale CCHP system.

2. Description of the micro-CCHP system

The choice of cogeneration configuration depends on the type of cogeneration cycle, power and heat requirements, and purpose and techniques of waste-heat utilization. The main components of a cogeneration system are the prime mover-generator set, heat recovery equipment, control equipment, and electrical transmission and distribution system. Various small-scale micro-CCHP systems, such as internal combustion engine based systems, fuel cell based

systems, Stirling engine based systems, micro-turbine based systems and etc., have been supposed. Fig. 1 shows the micro-CCHP system driven by internal combustion engine. The fuel, LPG or natural gas, is used in the engine to generate power. The engine jacket cooling water passes through the heat-recovery heat exchanger and is reheated by the exhausted gases, and then passes through an adsorption chiller to produce chilled water in summer or heat exchanger I to produce heating water in winter, after which the jacket water passes through heat exchanger II to produce domestic hot water. Then the jacket water returns to the engine. Finally, the produced chilled water, heating water and domestic hot water are supplied to the existing networks. The co-generated chilled water or heating water can be used directly with the existing air conditioning system.

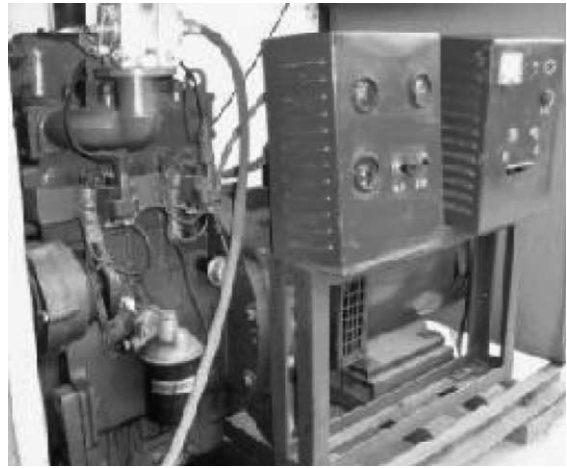


Fig. 2. Photograph of gas engine driven generator.

3. Description of the test facility

3.1. Characteristics of the gas engine driven generator and adsorption chiller

The engine is a double-cylinder, four-stroke, water-cooled, LPG and natural gas engine. The gas engine driven generator is shown in Fig. 2. The generator feeds 400/230 V, 50 Hz, three phase, four wire loads with power rating of 12 kW. The exhausted gases of combustion has a temperature of 580 °C at rated power. The noise rating of the generator set at 1 m with attenuation is lower than 60 dBA. For the generator set at rated power, its generating efficiency is 21.4%, and the heat recovered from the exhausted gases and from cylinder jacket cooling are 13.6 kW (24.3%) and 14.4 kW (25.7%), respectively.

The simple but effective adsorption chiller is shown in Fig. 3. This chiller can be regarded as one combined by two single-bed systems. Silica gel–water is selected as working pair in the new adsorption chiller special suited to use low temperature heat source. Water is taken as the refrigerant. This chiller is combined by three vacuum cavities: two desorption/adsorption working chambers and one heat pipe-

working chamber. There is only one vacuum valve installed to execute mass recovery process between the two desorption/adsorption working chambers. The operating reliability of the chiller is very good. The adsorber is one compact tube-fin heat exchanger, the condenser is tube-shell heat exchanger and the evaporator cooling is output through the methanol chamber, which acts like gravity heat pipe. In each adsorber 52 kg silica gel is filled. The mass of the refrigerant in each side is 12 kg, and that of the methanol is 9 kg. The heating/cooling time is 900 s, the mass recovery time is 180 s and the heat recovery time is 60 s. Table 1 shows the operating parameters of the small-scale adsorption chiller.

3.2. Experimental system

It is realized that the potential of the cogeneration system is dependent on the utilization of cogeneration product. In reality, it is difficult to match the produced load with existing requirements, especially in case of residential and commercial buildings, where it is more complicated because of limited end-use, only for heating, ventilation and air

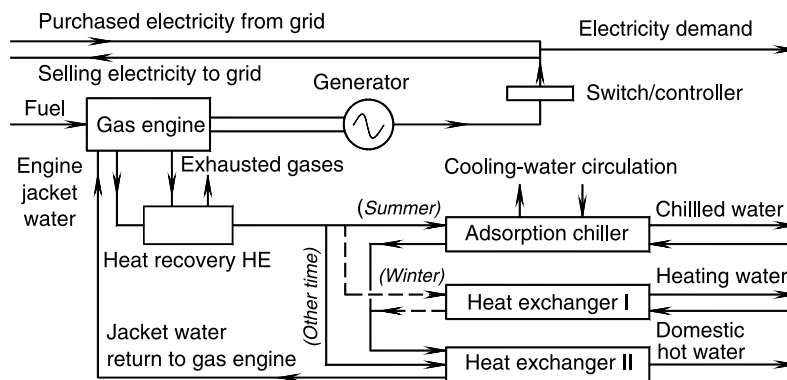


Fig. 1. Schematic diagram of micro-CCHP configuration.



Fig. 3. An effective Silica gel–water adsorption chiller.

conditioning (HVAC) systems. So the assessment of a micro-CCHP system and its match with electrical and thermal requirements in a domestic and commercial application is a crucial point. To this aim, a test facility was set up to supply better test-rig platform for cooling, heating and power cogeneration. The test facility is shown in Fig. 4. A small-scale adsorption chiller and a gas engine are used as part of the micro-CCHP system. The fan coils, cooling tower and electric lamp unit equipment are arranged to permit the simulation of cooling, heating and electric loads. A data acquisition system and measurement equipment is installed to measure the performance of the micro-CCHP system directly. Referring to the output rating of the micro-CCHP system's equipments, the experimental system has been planned to follow the cooling demand in the range 0–10 kW, the heating demand in the range 0–28 kW and the electric demand in the range 0–12 kW.

The operation principal of the micro-CCHP system is described as follows. At first, the gas engine is driven to supply electric power, and then the recovered waste heat from the cylinders cooling and exhausted gases are used to drive the small-scale adsorption chiller for cooling purpose, or, to output heating or hot water.

Table 1
Parameters and performance data of the small-scale adsorption chiller

Heat source water temperature (°C)	60–95
Heat source water flow rate ($\text{m}^3 \text{h}^{-1}$)	4.0
Chilled water inlet/outlet temperature (°C)	16/11
Cooling water inlet/outlet temperature (°C)	32/38
Chilled/cooling water flow rate ($\text{m}^3 \text{h}^{-1}$)	1.8/5.0
Performance	
Cooling capacity (kW)	6–10
COP	0.3–0.4

The cooling power produced by the adsorption chiller is transferred to an air-conditioning room by fan coil units. The heating load recovered from the circulated water by plate HE is transferred to the cooling tower that simulates heat users. The electrical load has been simulated by the use of electric lamps, in which 60 electric lamps (rated power 200 W each) are arranged in parallel connection, and they are divided into five groups and the electric power are 600, 1200, 2400, 3000 and 4800 W, respectively. Each group has independent switch, so the range of electrical loads is from 600 W to 12 kW.

The jacket water outlet temperature of the engine should not be higher than 95 °C, which is guaranteed by the electric switch valve 1. If the temperature is higher than 95 °C, the valve will be shut off for protection of the engine. In view of higher efficiency of gas engine, the jacket water inlet temperature had better be within the range of 60–85 °C. The temperature is kept within limited value by adjusting the opening of electric valves 2 and 4. The heat recovered from the exhausted gases can be regulated by the opening of the three-way valve 3 for the exhausted gases.

The analysis of the micro-CCHP system performance under consideration is based on energy balances on the whole system and internal devices, so the following measurements are performed:

- fuel gas temperature, pressure and volumetric flow rate;
- air volumetric flow rate to engine;
- exhausted gases temperature and volumetric flow rate;
- water volumetric flow rate and temperature at the inlet and outlet of the unit;
- electric voltage, frequency and power.

The following sensors and instruments are used:

- Platinum resistance temperature sensors (Pt100, grade A, $100 \Omega \pm 0.10\%$ at 0 °C) are used for water and fuel gas temperature measurements.
- Turbine flow meters ($\pm 0.5\%$ accuracy in a range of 0–10 $\text{m}^3 \text{h}^{-1}$, with 4–20 mA output) are used for water volumetric flow rate measurement.
- Vortex streets flow meter ($\pm 1.0\%$ accuracy in a range of 18–150 $\text{m}^3 \text{h}^{-1}$, with 4–20 mA output) is used for air flow rate measurement.
- Metallic flowrator flow meter ($\pm 1.5\%$ accuracy in a

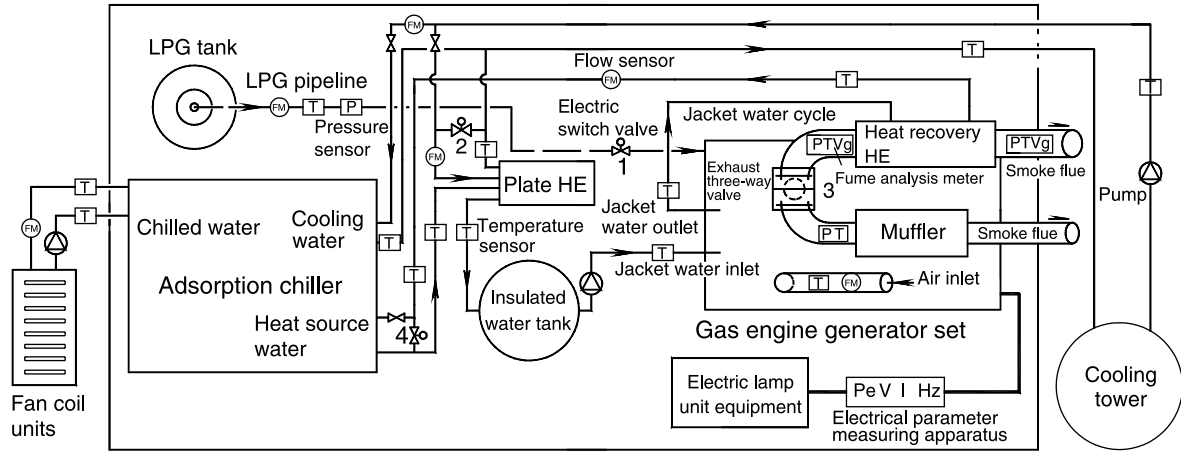


Fig. 4. Test facility layout (T : temperature sensor; P : pressure sensor; FM: flow sensor; Pe: electric power; V : voltage; I : current; Hz: frequency; V_g : gas volume flow).

range of $0.3\text{--}3\text{ m}^3\text{ h}^{-1}$, with $4\text{--}20\text{ mA}$ output) is used for fuel gas flow rate measurement.

- Fume analysis meter ($\pm 3.0\%$ accuracy each) is used for temperature and volumetric flow rate measurements of the exhausted gases.
- Electrical parameter measuring apparatus ($\pm 0.25\%$ accuracy) is used for electrical energy output measurement, including voltage, power and frequency.

A data acquisition logger reads and stores data about every 10 s. Fig. 5 shows the arrangement of the test facility.

3.3. Experimental methods and performance parameters

The test facility is used to test the micro-CCHP system performance at design load and part load conditions. These ‘field test’ results will be used to develop and verify the energy optimization model for the system and then optimize the micro-CCHP design and operation both from energetic and economic point of view. The experimental work is done under different working conditions. All the data in this paper



Fig. 5. Test facility view.

are obtained after the working condition is stable. The data of calculation parameters are the average values in one cycle of the adsorption chiller.

The performance parameters are calculated as follows:

$$Q_{\text{jacket_water_HE}}$$

$$= C_{p,w} m_{\text{hot}} (T_{\text{jacket_water_HE,out}} - T_{\text{jacket_water_HE,in}}) \quad (1)$$

where $Q_{\text{jacket_water_HE}}$ is recovered heat from cooling jacket, $C_{p,w}$ is specific heat of water, m_{hot} is hot water mass flow, $T_{\text{jacket_water_HE}}$ is water temperature of jacket heat exchanger, in and out represent inlet and outlet, respectively.

$$Q_{\text{exhausted_gases_HE}}$$

$$= C_{p,w} m_{\text{hot}} (T_{\text{exhausted_gases_HE,out}} - T_{\text{exhausted_gases_HE,in}}) \quad (2)$$

where $Q_{\text{exhausted_gases_HE}}$ is recovered heat from exhausted gases, $T_{\text{exhausted_gases_HE}}$ is water temperature of exhausted gases heat exchanger.

$$r_{\text{el_th}} = \frac{P_{\text{el}}}{Q_{\text{jacket_water_HE}} + Q_{\text{exhausted_gases_HE}}} \quad (3)$$

here $r_{\text{el_th}}$ is electricity to thermal heat recovered ratio, P_{el} is electric power output. The electrical efficiency, η_{el} is

$$\eta_{\text{el}} = \frac{P_{\text{el}}}{Q_{\text{LPG}}} \quad (4)$$

where LPG is liquefied petroleum gas, Q_{LPG} is its energy input based on lower calorific value (46.5 MJ/kg). The thermal efficiency, η_{th} is

$$\eta_{\text{th}} = \frac{Q_{\text{jacket_water_HE}} + Q_{\text{exhausted_gases_HE}}}{Q_{\text{LPG}}} \quad (5)$$

The total thermal and electrical efficiency, η_{total} is

$$\eta_{\text{total}} = \eta_{\text{el}} + \eta_{\text{th}} \quad (6)$$

The refrigeration power of the adsorption chiller is

$$Q_{\text{refrigeration_power}} = C_{p,w} m_{\text{chilled}} (T_{\text{chilled,in}} - T_{\text{chilled,out}}) \quad (7)$$

where m_{chilled} is chilled water mass flow, T_{chilled} is chilled water temperature. The heating power to drive adsorption chiller is

$$Q_{\text{ad_chiller_heating_power}} = C_{p,w} m_{\text{heat_source}} (T_{\text{heat_source,in}} - T_{\text{heat_source,out}}) \quad (8)$$

here $m_{\text{heat_source}}$ is hot water mass flow of adsorption chiller, $T_{\text{heat_source}}$ is hot water temperature to adsorption chiller. The refrigeration COP of the adsorption chiller is defined as

$$\text{COP}_{\text{ad_chiller}} = \frac{Q_{\text{refrigeration_power}}}{Q_{\text{ad_chiller_heating_power}}} \quad (9)$$

The primary energy rate PER is then

$$\text{PER}_{\text{CHP}} = \frac{Q_{\text{jacket_water_HE}} + Q_{\text{exhausted_gases_HE}} + P_{\text{el}}}{Q_{\text{LPG}}} \quad (10)$$

$$\text{PER}_{\text{CCP}} = \frac{Q_{\text{refrigeration_power}} + P_{\text{el}}}{Q_{\text{LPG}}} \quad (11)$$

$$\text{PER}_{\text{CCHP}} = \frac{Q_{\text{heat_load}} + Q_{\text{refrigeration_power}} + P_{\text{el}}}{Q_{\text{LPG}}} \quad (12)$$

where the subscripts CHP is combined heat and power, CCP is combined cooling and power, CCHP is combined cooling, heating and power, respectively.

If HP is heat and power supplied separately, CP is cooling and power supplied separately, CHPS is cooling, heat and power supplied separately, conv is conventional independent system, then the respective PER are

$$\text{PER}_{\text{HP_conv}} = \frac{Q_{\text{jacket_water_HE}} + Q_{\text{exhausted_gases_HE}} + P_{\text{el}}}{(Q_{\text{jacket_water_HE}} + Q_{\text{exhausted_gases_HE}})/\eta_{\text{boiler_conv}} + P_{\text{el}}/\eta_{\text{el_conv}}} \quad (13)$$

$$\text{PER}_{\text{CP_conv}} = \frac{Q_{\text{refrigeration_power}} + P_{\text{el}}}{Q_{\text{refrigeration_power}}/(\text{COP}_{\text{el_chiller_conv}} \eta_{\text{el_conv}}) + P_{\text{el}}/\eta_{\text{el_conv}}} \quad (14)$$

$$\text{PER}_{\text{CHPS_conv}} = \frac{Q_{\text{heat_load}} + Q_{\text{refrigeration_power}} + P_{\text{el}}}{Q_{\text{heat_load}}/\eta_{\text{boiler_conv}} + Q_{\text{refrigeration_power}}/(\text{COP}_{\text{el_chiller_conv}} \eta_{\text{el_conv}}) + P_{\text{el}}/\eta_{\text{el_conv}}} \quad (15)$$

where $\eta_{\text{boiler_conv}}$, $\eta_{\text{el_conv}}$, $\text{COP}_{\text{el_chiller_conv}}$ are the boiler efficiency, electrical efficiency and the COP of electrical air conditioner of the conventional independent system.

The ratio of the heat load supplied to recovered heat is

$$\theta = \frac{Q_{\text{heat_load}}}{Q_{\text{jacket_water_HE}} + Q_{\text{exhausted_gases_HE}}} \quad (16)$$

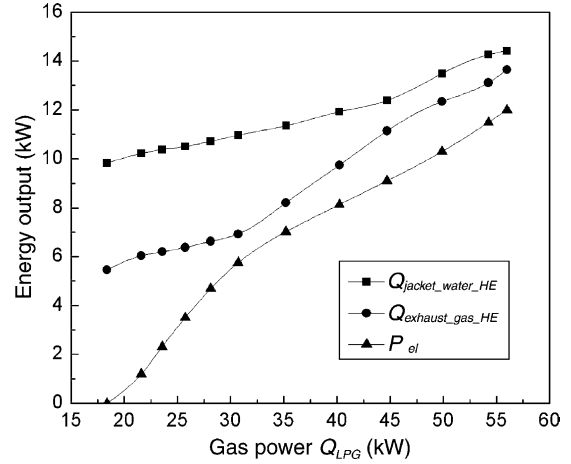


Fig. 6. Thermal and electrical outputs against gas power.

where $Q_{\text{heat_load}}$ is heat load supplied by the micro CCHP system.

4. Experimental results

A comprehensive database of system conditions was obtained from the test program that allowed an analysis of the system performance to be made. The main results are shown in Figs. 6–15 and are discussed in Sections 4.1–4.4.

4.1. Thermal and electrical performance

The effect of the thermal and electrical performance of variations in the gas power of generator set is shown in Figs. 6–9. Fig. 6 shows thermal heat recovered from jacket water and exhausted gases and electrical output of the system as function of gas power. It can be stated that as the gas power of the generator increases, thermal heat recovered from jacket water and exhausted gases also increases, as does electrical output.

With the increase of the LPG burning power, the growing gradient of electrical output curve is greater than those of thermal curves until the gas power reaches up to 31 kW, in which the difference of three curves' gradient is becoming smaller. It is seen that the thermal heat from

jacket water is the most, and the electrical output is the least among the above three kinds of energy outputs. The gas engine works at a constant rotational speed of 1500 rpm. At an electrical power of $P_{\text{el}}=0$ kW, a minimum thermal heat recovered from jacket water and exhausted gases is 15.9 kW. At an electrical power of $P_{\text{el}}=12$ kW, a maximum thermal heat of 28.1 kW is produced. This means that

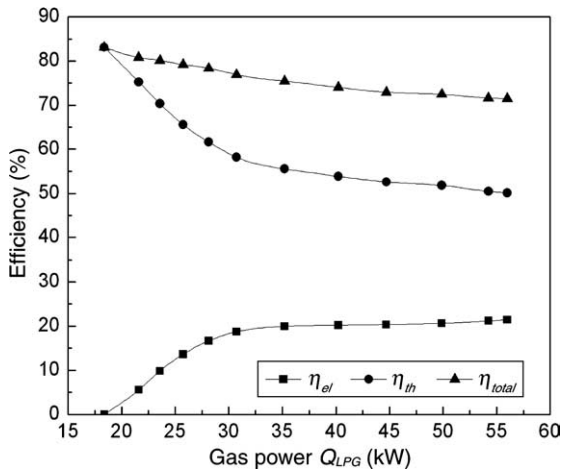


Fig. 7. Thermal and electrical efficiencies against gas power.

thermal load of generator set can only be regulated in a range of 56.6–100%. Thermal, electrical and overall efficiencies of the system are shown as function of gas power in Fig. 7. As the gas power increases, electrical efficiency increases, while thermal and overall efficiencies decrease. Opposed behavior of electrical efficiency to thermal efficiency can be explained by the evidence that the exhausted gases temperature is rising with engine output power increasing. The tests also showed that if the gas power is higher than 35 kW, the variations of the thermal, electrical and overall efficiencies are very small. Fig. 8 shows that exhausted gases inlet temperature of heat recovery heat exchanger increases rapidly as the electricity output of generator increases, and the corresponding output temperature also increases but variations are smaller. At an electrical power of $P_{el}=12$ kW, exhausted gases inlet and outlet temperatures of that are about 600 and 116 °C, respectively.

Fig. 9 shows the higher gas power to the generator causes higher electricity to thermal heat recovered ratio. It also

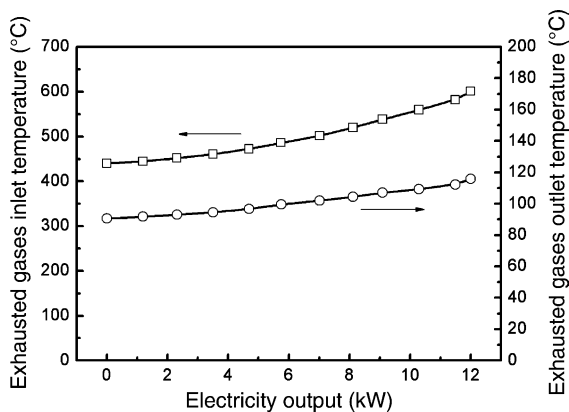


Fig. 8. Exhausted gases inlet and outlet temperature of heat recovery heat exchanger against electric power of the generator set.

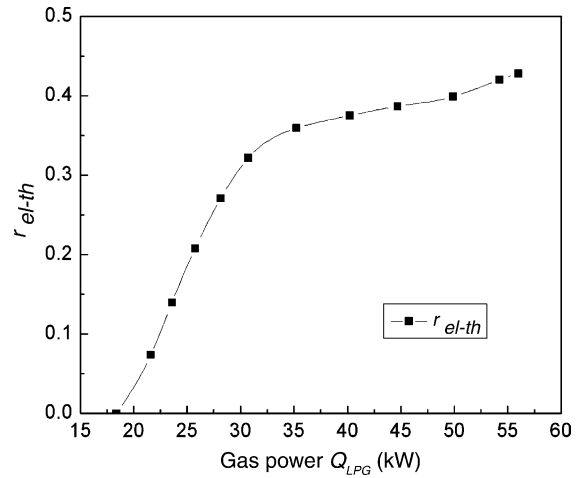


Fig. 9. Electricity to thermal heat recovered ratio against gas power.

illustrates that the ratio remains smaller changes until the gas power is below 35 kW when it begins to decrease rapidly. At maximum gas power, the ratio is at around 0.43. Other internal combustion engine driven generators claim to reach a ratio of 0.5–1, as their electrical efficiencies are much higher [22].

4.2. Adsorption chiller performance

The heat-source water inlet temperature is much critical to the performance of the adsorption chiller, as shown in Fig. 10. It shows that as hot water inlet temperature increases, refrigeration power and coefficient of performance (COP) also increase. The effect of the chilled water inlet temperature on refrigeration power and COP is also shown in Fig. 10. Increasing the chilled water inlet temperature is advantageous to improve the performance of the system. The flow rates of hot water, cooling water and chilled water were kept constant, which are 4.02, 3.86 and

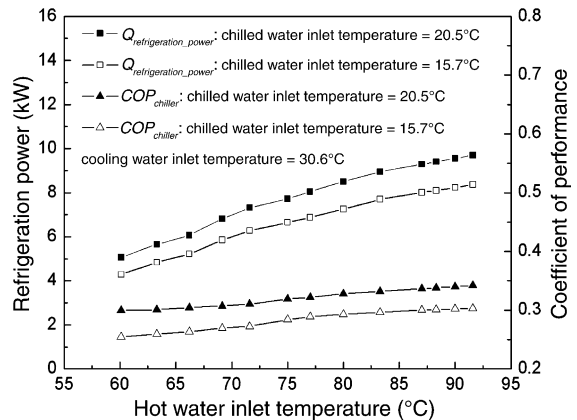


Fig. 10. Refrigeration power and COP variations with heat-source water temperature.

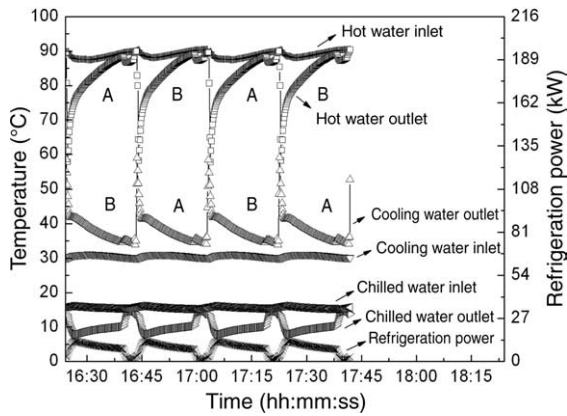


Fig. 11. Heat-source water, cooling water, chilled water and refrigeration power against operating time of adsorption chiller.

$1.31 \text{ m}^3 \text{ h}^{-1}$, respectively. The tests show that with hot water inlet temperature increasing in the range of 60.1 – 91.6°C , the refrigeration power is increasing from a minimum 5.1 kW to a maximum 9.7 kW and the COP is increasing from 0.3 to 0.34 when the chilled water inlet temperature is 20.5°C and the cooling water inlet temperature is 30.6°C . At the same heat-source water inlet temperature and the same cooling water inlet temperature, when the chilled water inlet temperature increasing from 15.7 to 20.5°C , more than 13.7% improvement of refrigerating power and more than 10.2% improvement of COP have been obtained.

Fig. 11 shows the temperature variations of hot water (as heat-source), cooling water, chilled water and refrigeration power with operating time during two cycles. One whole refrigeration cycle of adsorption chiller consists of six working processes, which can be described by an example of the first cycle in Fig. 11. Adsorber A desorption–adsorber B adsorption process ($16:24:10$ – $16:39:10$), mass recovery

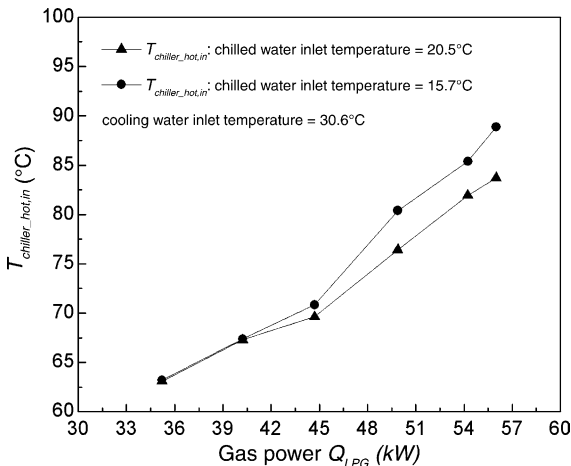


Fig. 12. Adsorption chiller heat-source water inlet temperature against gas power of the micro-CCHP system.

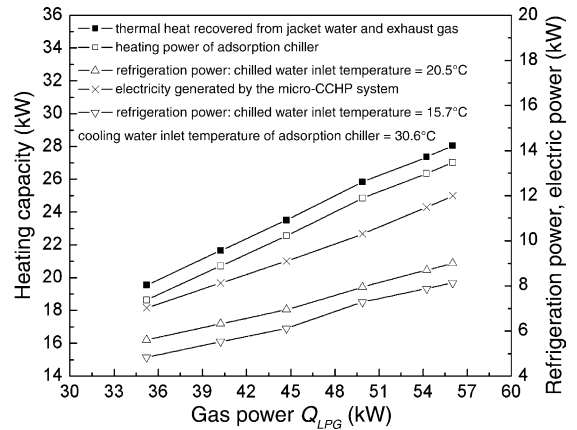


Fig. 13. Thermal recovered and energy output of CCP against gas power of the micro-CCHP system.

process from adsorber A to B ($16:39:10$ – $16:42:10$), heat recovery process from adsorber A to B ($16:42:10$ – $16:43:10$), adsorber B desorption–adsorber A adsorption process ($16:43:10$ – $16:58:10$), mass recovery process from adsorber B to A ($16:58:10$ – $17:01:10$) and heat recovery process from adsorber B to A ($17:01:10$ – $17:02:10$).

In these two cycles, the hot water (as heat-source) inlet temperature has a little change of only 3°C at approximately 88.9°C . The inlet temperatures of cooling water and chilled water are controlled to change a little, which are kept close to 30.6 and 15.7°C , respectively. The outlet temperatures of hot water, cooling water and chilled water changes largely. The differences of maximum and minimum values of them are at about 32.6 , 25.2 and 6.9°C , respectively. The refrigeration power varies from 1 to 12.4 kW . In desorption/adsorption process, chilled water outlet temperature decreases quickly to the lowest temperature and then increases slowly. In the mass recovery process, the hot water outlet temperature will decrease firstly and then increase, while the cooling water outlet temperature

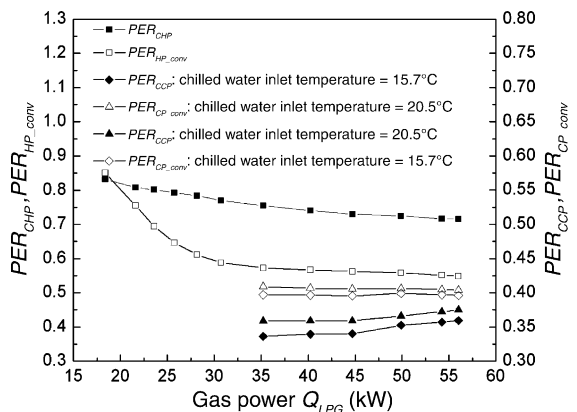


Fig. 14. PER of the micro-CCHP system and conventional independent system against gas power.

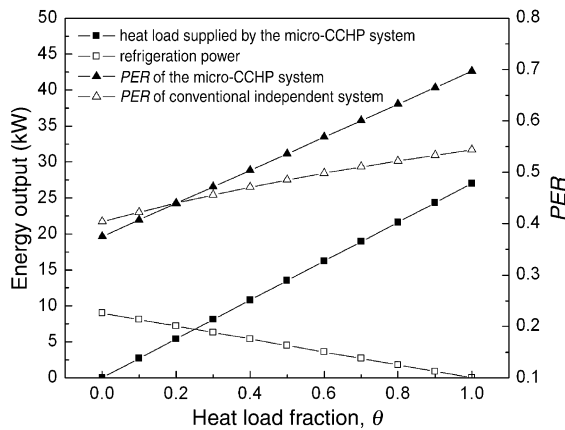


Fig. 15. Energy output and PER of the micro-CCHP system and conventional independent system against heat load fraction.

increases firstly and then decreases. This is due to the variations of the pressure difference between the adsorber and the desorber.

4.3. Combined cooling and power (CCP) performance

Due to the limit of the jacket water inlet temperature of the gas engine, it had better not be below 60 °C. This causes that the chiller inlet temperature should be higher than 63 °C. The whole thermal heat recovered is supplied to the adsorption chiller. Some combined cooling and power performance of the micro-CCHP system is shown in Figs. 12 and 13.

The hot water inlet temperature of the adsorption chiller increases as the gas power of the micro-CCHP system does, as shown in Fig. 12. It also shows that the lower chilled water inlet temperature of the adsorption chiller causes an increase of the hot water inlet temperature with the same gas power. The reason is that the lower chilled water inlet temperature causes the lower evaporation pressure in general adsorption cycle. When the evaporation pressure becomes lower, the circulating adsorption capacity of adsorbent decreases. As a result, the heat capacity demand of adsorbent also decreases. In order to consume the same recovered waste heat from the gas engine, the adsorption chiller needs the higher hot water inlet temperature. When the generator runs at rated power, the heat-source water inlet temperatures are 83.7 and 88.9 °C at the chilled water inlet temperatures of 20.5 and 15.7 °C, respectively.

The tests also showed that the higher gas power caused higher refrigeration power, as shown in Fig. 13. At the chilled inlet temperature of 20.5 °C and the cooling water inlet temperature of 30.6 °C, the refrigeration power is 9 kW and electrical energy is 12 kW at rated gas power input of 56 kW of the micro-CCHP system. It is evident that the match of the gas engine and the small-scale adsorption chiller is reasonable. The maximum recovered heat from the gas engine is near to the rated driving heat power of the

adsorption chiller. So the system can supply 12 kW electrical power and 9 kW refrigeration power with high efficiency.

4.4. Energetic evaluation of the micro-CCHP system in comparison to conventional independent systems

In the conventional independent system, the electricity demand is met by purchase of electricity from grid, the cooling demand is met by electrical air conditioner and the heating demand is met by the central heating and hot water supply system. The decisive value for energy efficiency evaluation of combined cooling, heating and power system is the primary energy ratio (PER). The PER is the ratio of the required energy output to the primary energy demand, and consequently, the system with the highest value of PER is considered the best with regard to energy consumption [23–26]. Reference energy parameters of boiler efficiency, electrical efficiency and the COP of electrical room air conditioner of conventional independent system are stated for applications in China, which are 0.85, 0.3, and 2.5, respectively.

Fig. 14 shows the PER variations of the micro-CCHP and conventional independent system with different gas power. For the heat and power generation (CHP), the PER of the micro-CCHP system is higher than one of conventional independent system at the same gas power except the minimum point. In the case of the minimum gas power, the system works like a boiler with thermal efficiency of 83.1%. The PER difference between two systems remains fairly constant when gas power is higher than 35.2 kW. At the maximum gas power, the difference value is about 0.167.

For the cooling and power generation (CCP), the PER of the micro-CCHP system is lower than one of conventional independent system at the same gas power. This is due to the lower COP of adsorption chiller. As the gas power increases, the PER difference between two systems decreases. For the chilled water inlet temperatures of 15.7 and 20.5 °C, the PER differences are 0.036 and 0.029, respectively, at the maximum gas power. As the gas power is higher than 44.7 kW, the PER difference begins to decrease rapidly. This is because in this case, the COP of adsorption chiller has a rapid increase.

When the micro-CCHP system supplies cooling, heating and electrical power simultaneously (CCHP), the total PER will be between one of the corresponding CCP and CHP of the micro-CCHP system. In view of energy saving, because the PER of CHP is higher than that of CCP for the micro-CCHP system, it is advantageous that the system supplies more heat load to users, as shown in Fig. 15. It shows the PER variations of two systems against heat load fraction at the maximum gas power of the micro-CCHP system with electricity load generated of 12 kW. It is evident that the PER of two systems increases when the heat load fraction increases. As the ratio is higher than 0.2, the PER of the micro-CCHP system is higher than that of the conventional

independent system. At the ratio of 0.5, the micro-CCHP system supplies heat load of 14.1 kW and cooling load of 4.7 kW. In this case, the PERs of the micro-CCHP system and conventional system are 0.55 and 0.50, respectively. This means the micro-CCHP system saves 9.1% of the primary energy compared to the conventional independent system. The maximum saving of the primary energy is 23.3% when the thermal heat recovered is used to supply heat load completely. In other words, the heat load fraction is equal to 1.

5. Conclusions

In this study, the micro-CCHP system and test facility equipped with a small LPG and natural gas engine and a new adsorption chiller are described. The tests conducted provided a comprehensive database of the operating parameters for the micro-CCHP system. The system can supply electricity load of 12 kW and heat load of 28 kW or cooling load of 9 kW simultaneously.

In comparison to large-scale CCHP system, the small-scale CCHP test facility supplies better test-rig platform for cooling, heating and power cogeneration. Sufficient researches on energy management and operational optimization of CCHP systems could be made by using it. The electricity load, heat load and cooling load of the micro-CCHP system can be regulated according to the requirement. The PER of CHP of the micro-CCHP system is much higher than that of CCP of the system. This means that the higher heat load supplied causes higher PER of the micro-CCHP system. In this case, the micro-CCHP system can save more primary energy than the conventional independent system. The information gathered will be very useful in a future study of the system's energy and economic optimization and the use of the micro-CCHP system in small-scale users, such as residential and commercial ones.

Acknowledgements

This work was supported by the state Key Fundamental Research Program under the contract No. G2000026309, the National Science Fund for Distinguished Young Scholars of China under the contract No. 50225621, and also the Research Fund for the Doctoral Program of Higher Education under contract No. 20040248055.

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